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Abstract

This thesis paper proposes a simplified analysis method for the complex three-dimension continuum body by the first order analysis (FOA). For a complicate continuum body it is difficult to understand how the load is transmitted to the supports and the influence of design variable changes to the total deformation or the stress concentration factor. To overcome these difficulties, a simplified finite element composed by beam elements is suggested for the deformation analysis of 3D–bodies, and to predict the stress concentration factor, a formulation of stress concentrated members for the FOA has been proposed. When a complicate solid structure is simplified as an assemblage of simple beam elements applied to element forces such as axial force, shear force, bending moment and twisting moment, the total stiffness equation for the FOA of the solid structure is constructed and solved for the given boundary and loading conditions by the usual manner. Then, the total deformation of simplified model is estimated by the simplified model and the element forces are calculated in the element level. The stress concentration of simplified each member with holes, notches and variable cross section is also predicted by the suggested approximate formulation.

1. Introduction

Computer-Aided Engineering (CAE) by Lemon et al.[1] has been widely accepted by the industrial designers and the simulation engineers. And the structural optimization software such as GENESIS enrolled an important contribution for the design optimization. However, it is difficult for the designers to understand, why the final design obtained by the optimization software is optimum, as skillful engineering know-how. To overcome this problem, a new concept of design tool named the First Order Analysis (FOA) has been proposed by Nishigaki et al.[2],[3] to treat the skeletal structures which consists of beams and panels. The developed FOA tool provides a useful aid to design the load transmission paths and topology optimization of skeletal structures at upstream stage of design processes, because the tool includes a variety of graphical interfaces using MS Excel for the design engineers and powerful topology optimization module.

However, the FOA proposed by Nishigaki et al.[(2), (3)] includes only the beam and panel elements as design elements, it is not enough to simplify the design model of complicate general solid structures such as machine tools in all kinds of structural design optimization. Moreover, the suggested FOA can treat only the rigidity of the body structure. On the other hand, some researches have also been tried to construct the low-fidelity CAE models to reduce the computational cost for the optimization of the complicate structural design of high-fidelity models[(4),(5)]. However, it is difficult to establish a unified theory and manner to construct low-fidelity models for the complicate 3D-bodies.

When we consider the optimum design of continuum structures in the upstream design process, the stress concentration as well as the structural rigidity should be taken in to account in the design process and the FOA should be extended to provide such ability for the designers. The objective of this research is to develop a new FOA concept, which is able:

(1) to predict total deformation and load transmission as well as stress concentration of complicate structures by the simplified model, and

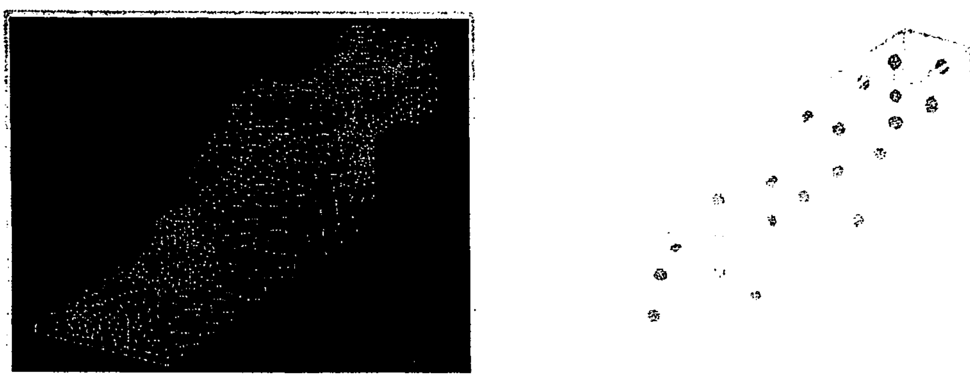
(2) to assist the design engineers to optimize the structure, to make them understandable how the load is transmitted rationally to the support, and which parameter is important for optimization of structures.

At first, a simplified finite element that can express tensile, compressive, torsion and bending deformations of the body by extending simple beam element is proposed. Then, the proposed FOA method is applied to a couple of basic analysis examples as well as complicate 3D-bodies.And To develop formulas of stress concentration factor prediction, we select a U-notch in cylinder as an example among the different many kinds of element shapes, which is applied tensile or bending or shear loading separately at the ends. The formulation will be shown in the next section. By using the same way we can prepare formulas for other shape of the model like plate with other kind of notch or hole or without any defects. In the practical cases, the design engineers are expected creative design products, and they have to check the possibility of many kinds of structural candidates of very complicate machine elements. To realize this design process and to assist them for creative and interactive thinking on the computer, software embedded the following mentioned steps with graphical interface is expected to be developed. To achieve such kind of design environment for the engineer, it is required to prepare the element stiffness library and stress concentration factor library as functions of the design parameters and the element forces for the many kinds of elements with many kinds of notches, defects, holes and variable cross sections.

2. Formulation of Simplified 3D-FOA Element

2.1 FOA of Three-dimensional Body

The main themes of 3D-body FOA are: (1) to supply analysis methodology of low-fidelity model of complicate 3D-bodies, (2) to prepare some graphic interfaces for designers using such as Microsoft/Excel[(2), (3), (6)] for the design assistance and optimization, and (3) to offer interactive design function to show the effects such as the design parameter changes and load transmitting paths. Figure 1 shows an image of 3D-body FOA by comparing with the fine CAE model.



(a) Finite element mesh for CAE　　(b) Descritization for FOA

Fig.1 Comparison of FOA model and FEM

2.2 Construction of FOA Model for Continuum Body

For a complicate continuum structure it is difficult to understand directly how the load is transmitted to the supports, and the influence of design variable changes to the total deformation or the stress concentration factor. So to satisfy these requirements, a FOA formulation for the continuum structures is constructed by the following steps:

(1) Simplify the whole complex model as an assemblage of a various kinds of simple elements..

(2) Decompose assembled elements to a set of simple elements applied forces such as axial and shear forces, bending and twisting moments. If the relationship between the element stiffness and the design parameters is prepared for a various kinds of simplified elements in the computer library, we may invoke the stiffness matrix for the specified design parameters.

(3) Construct the total stiffness equation as an assemblage of the simple elements. For example, if a planar beam element with a V-notch has several design parameters such as width W, a half of the distance a between the two notch roots, root radius of notch ρ, and notch angle α as shown in Fig.2, the element stiffness equation is given such as

$$F^c = K^e U^e \tag{1}$$

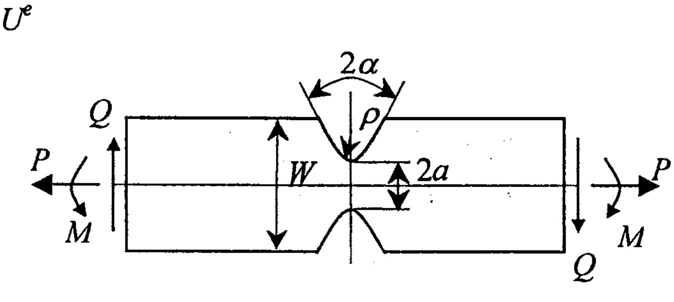


Fig. 2　A V-notch element with different kind of load.

where $F^e = (P_1, Q_1, M_1, P_2, Q_2, M_2)^T$, $U^e = (u_1, v_1, \theta_1, u_2, v_2, \theta_2)^T$ when P_i, Q_i, $M_i (i = 1, 2)$ denote axial force, shear force and bending moment, and u_i, v_i, $\theta_i (i = 1, 2)$ denote the axial and vertical displacements, and rotation of both ends, respectively. The stiffness matrix of simplified element should be prepared as a function of the design parameters as $K^e = K^e(\rho, a, W, \alpha)$ in advance. Then, the element stiffness matrix is assembled to obtain the total stiffness matrix, and thereby solved for the total displacement of the structure under the given loading and boundary conditions as usual manner of the finite element analysis. Then, the structural deformation of whole structure is predicted roughly.

(4) Calculate the element forces to predict the stress concentration for the given design parameters. To do so, some sophisticated formulas are prepared for the maximum stress for the typical element including design parameters.

(5) Display the deformation and element forces. Then, the designer can observe how the load is transmitted, and also how the stress concentrates at each element.

(6) Change the structural topology if needed and optimize the design parameters.

When we have to change any dimension or shape, we may go to step 2 or to step 1 if required the structural model change.

2.3 Derivation of Simplified Finite Element Stiffness

When we want to obtain precise stiffness of 3D-body, the analysis model has to be descritized into fine meshes by the usual finite elements. On the other hand, the continuum body modeled by an equivalent beam element that transmits only tension/compression, torsion and bending is enough to estimate roughly the deformation of complicate 3D-bodies and load transmitting path. To develop such kind of simplified finite element, it is assumed that an approximate deformation mode of complex continuum body is expressed by the combination of tensile, compressive, bending and torsional basic deformations. As the simplest approximation example of 3D-body deformation, suppose that a general hexahedral block shown in Fig.3(a) is in a state of uniform deformation. The hexahedral block consists of seven representative nodes, six nodes of which are located on the center point of hexahedral edge surfaces, and the 7^{th} node of which is located at the center of hexahedron. The distances L_x, L_y and L_z in Fig.3(b) denote lengths between node 1 and node 2, nodes 3,5 and nodes 4,6. Each node has 6 degrees of freedom of displacement (u_i, v_i, w_i, θ_{xi}, θ_{yi}, θ_{zi}) ($i = 1, 2, 3\ldots, 7$) in the local coordinate systems (x', y', z') of an element. Corresponding to the displacement components, axial forces (Q_{xi}, Q_{yi}, Q_{zi}), and moments (M_{xi}, M_{yi}, M_{zi}) are considered at each node. When we regard a hexahedron as an assembled structure of six beams which consist of elements connecting the nodes at the center of hexahedral edge surfaces and a center node of hexahedron, the stiffness of hexahedron is derived by differentiating the total potential energy π of the hexahedron with respect to each displacement component(7)(8).

Let us consider the total potential energy π of simple deformation such as tension/compression, bending and torsion under Bernoulli-Euler beam theory. When a beam consists of nodes 1 and 7 which take the deflection and the rotation as nodal unknown variables separately, the stiffness equation in the axial direction is derived by assuming the linear interpolating function as

$$\frac{\partial \pi}{\partial u_1} = \frac{EA}{L_x/2}(u_1 - u_7) - Q_{x1} = 0 \tag{2}$$

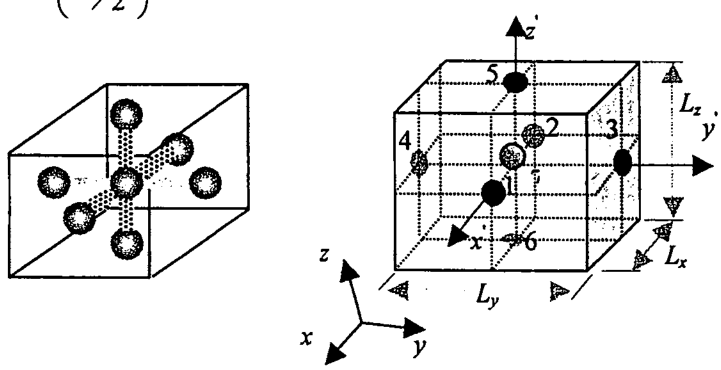
By considering the potential energy for bending and assuming Hermite shape functions for the deflection to keep the conforming condition, we can derive stiffness equation as follows;

$$\frac{\partial \pi}{\partial w_1} = \frac{12EI_y}{\left(L_x/2\right)^3}(w_1 - w_7) + \frac{6EI_y}{\left(L_x/2\right)^2}\left(\theta_{y1} + \theta_{y7}\right) - Q_{z1} = 0$$

$$\frac{\partial \pi}{\partial \theta_{y1}} = \frac{6EI_y}{\left(L_x/2\right)^2}(w_1 - w_7) + \frac{2EI_y}{\left(L_x/2\right)}\left(2\theta_{y1} + \theta_{y7}\right) - M_{y1} = 0 \tag{3}$$

and for the torsional stiffness equation,

$$\frac{\partial \pi}{\partial \theta_{x1}} = \frac{C}{\left(L_x/2\right)}(\theta_{x1} - \theta_{x7}) - M_{x1} = 0 \tag{4}$$



(a) Hexahedral block (b) Local coordinates and nodes

Fig.3 Simplified structural element

where, E and A are the Young's modulus and the cross-sectional area of the beam, I_y is the second moment of inertia, and C is the torsional rigidity. By the same way, the stiffness equations of other 5 beams are also derived. These stiffness equations are expressed by the equilibrium equation in the matrix form of Eq.(1), in which K^e is the element stiffness matrix of 42 by 42 components for the simplified 3D-deformation analysis, where the proposed element stiffness does not take into account the influence of Poisson's ratio.

2.4 Derivation of Equivalent Cross-sectional Area, 2nd Moment of Inertia, Tortional Rigidity and Stiffness Matrix

(a) Equivalent Cross-Sectional Area (A_{eq})

When a load is applied on the end surface of regular quadrilateral block with uniform cross-section, the stiffness matrix can be calculated by the direct volume integration. However, for the non-uniform and irregular general shape as shown in Fig.4 we can not calculate the stiffness matrix directly by the usual formulas. Therefore, we have to consider more general shape by introducing coordinate transformation and numerical integration. Then, the equivalent stiffness matrix of general shape is derived by equating the strain energy of general solid model to the strain energy of simplified solid model with same volume.

Let us consider a block of general shape in Fig.4(a). When a load P is applied on an element surface in the longitudinal direction, the strain energy for uniform shape element of beam which consists of nodes 1 and 7 can be express as follows:

$$U_{eq} = \frac{EA_{eq-x}(u_1 - u_7)^2}{2L_x} \tag{5}$$

where $A_{eq\text{-}x}$, L_x and (u_1-u_7) are the equivalent cross-sectional area, the length of the element in the x-direction and displacement, respectively. On the other hand, in the general case the strain energy can be expressed as follows:

$$U_{general} = \frac{E(u_1 - u_7)^2}{2L_x^2} \int_0^{L} A(x)dx \tag{6}$$

In the practical evaluation, by introducing local coordinates, $-1 \le \xi, \eta, \zeta \le 1$ we can get a general form of integration as

$$I_{mn}(x(y,z)) = \int_{-1}^{1} \int_{-1}^{1} y^m z^n |J| d\eta d\zeta \Big|_{at\ x} \qquad (m, n = 0, 1, 2) \tag{7}$$

where

$$|J| = \begin{vmatrix} \frac{dy}{d\eta} & \frac{dz}{d\eta} \\ \frac{dy}{d\zeta} & \frac{dz}{d\zeta} \end{vmatrix} = \left(\frac{dy}{d\eta} \cdot \frac{dz}{d\zeta} - \frac{dz}{d\eta} \cdot \frac{dy}{d\zeta} \right)$$

For example, an equivalent cross-section is given by I_{00} when m=n=0, and equivalent second moment of inertia around y- and z-axis are given by I_{02} and I_{20} when m=0, n=2 and m=2, n=0. By using Gauss quadrature,

$$I_{mn}(x_i(y,z)) = \sum_{i=1}^{smp} \sum_{j=1}^{smp} w_i w_j y^m(\eta_i, \zeta_j) z^n(\eta_i, \zeta_j) |J(\eta_i, \zeta_j)| \Big|_{at\ x=\eta} \tag{8}$$

where smp and w_i, w_j denote the integration order and corresponding weights in each direction. Then, the equation of strain energy $U_{general}$ becomes as follows:

$$U_{general} = \left(\frac{L_x}{2} \right) \frac{E(u_1 - u_7)^2}{2L_x^2} \int_{-1}^{1} A(\xi) d\xi$$

$$\frac{E(u_1 - u_7)^2}{2L_x^2} \left(\frac{L_x}{2} \right) \sum_{i=1}^{smp} w_i A(\xi_i) = \frac{E(u_1 - u_7)^2}{2L_x^2} \left(\frac{L_x}{2} \right) \sum_{i=1}^{smp} w_i I_{00}(\xi_i) \tag{9}$$

By equating $U_{eq} = U_{general}$, then we can get A_{eq} as

$$A_{eq-x} = \frac{1}{2} \sum_{i=1}^{smp} w_i I_{00}(\xi_i) = \frac{1}{2} \sum_{i=1}^{smp} w_i \left[\sum_{j=1}^{smp} \sum_{k=1}^{smp} w_j w_k y^0(\eta_j, \zeta_k) z^0(\eta_j, \zeta_k) |J(\eta_j, \zeta_k)| \right] \tag{10}$$

(b) Equivalent Second Moment of Inertia (I_{eq})

Let us consider again a beam element which consists of two nodes. For the case of bending moment around the y-axis, when we assume cubic interpolating functions for the deflection by the terms of nodal defections and rotations around the axis as usual beam element, the strain energy for uniform shape is expressed as follows:

$$U_{eq} = \frac{EI_{eq-y}}{2L_x^3} (a^2 + ab + \frac{b^2}{3}) \tag{11}$$

where

$$a = -6(w_1 - w_7) - 2L_x(2\theta_{y_1} + \theta_{y7})$$
$$b = 12(w_1 - w_7) + 6L_x(\theta_{y_1} + \theta_{y7})$$

On the other hand, for general shape the strain energy can be expressed by the following equations:

$$U_{general} = \frac{E}{2L_x^3} \{ (\frac{a^2}{2} + \frac{ab}{2} + \frac{b^2}{8}) I^{(0)}_{eq-y} + (\frac{ab}{2} + \frac{b^2}{4}) I^{(1)}_{eq-y} + \frac{b^2}{8} I^{(2)}_{eq-y} \} \tag{12}$$

where

$$I^{(0)}_{eq-y} = \int_{-1}^{1} I_{20}(\xi) d\xi, \quad I^{(1)}_{eq-y} = \int_{-1}^{1} \xi I_{20}(\xi) d\xi, \quad I^{(2)}_{eq-y} = \int_{-1}^{1} \xi^2 I_{20}(\xi) d\xi$$

Unlike the calculation of $A_{eq\text{-}x}$, it is found that the equivalent second moment of inertia I_{eq} can not be decided directly by equating Eq.(11) and Eq.(12).

(c) Tortional Rigidity:

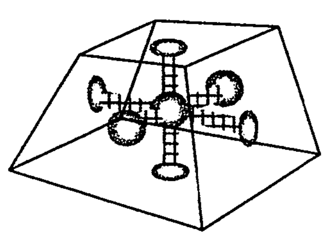
Again for a rectangular element, whose cross-sectional area is given as $l \times h$ (where $l \geq h$), let us assume that the 2nd moment of inertia equation by zero-order can be expressed as follows:

$$I_{eq-y} = \frac{lh^3}{12}, \quad I_{eq-z} = \frac{l^3 h}{12} \tag{13}$$

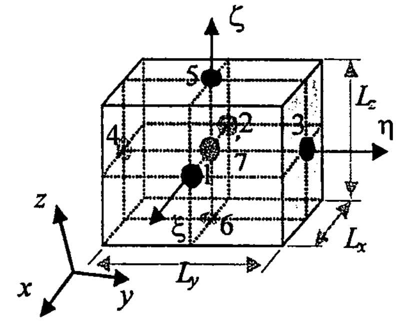
Now solving Eq.(13) we can easily get the equivalent values of l and h and by substituting these values into Eq.(14), the equivalent torsional rigidity, C_{eq}. is obtained[8] as follows:

$$C_{eq-x} = \left\{ \frac{1}{3} - 0.21 \frac{h}{l} \left(1 - \frac{h^4}{12 l^4} \right) \right\} l h^3 G \tag{14}$$

where G denotes the shear modulus.



(a) General element



(b) Local coordinates

Fig.4 General structural element and its coordinate transformation

(d) Stiffness Matrix:

After calculating A_{eq}, I_{eq}, and C_{eq}, when we adopt a conforming element which takes the deflection and rotation of nodes the unknown variables separately, finally we can derive the total stiffness matrix as follows:

$$\left[\begin{array}{cccccc|cccccc}
k_{11} & & & & & & & & & & & \\
 & k_{22} & & & & & & & & & & \\
 & & k_{33} & & & & & & & & & \\
 & & & k_{44} & & & & & \text{Sym.} & & & \\
 & & k_{53} & & k_{55} & & & & & & & \\
 & k_{62} & & & & k_{66} & & & & & & \\
\hline
k_{71} & & & & & & k_{77} & & & & & \\
 & k_{82} & & & & k_{86} & & k_{88} & & & & \\
 & & k_{93} & & k_{95} & & & & k_{99} & & & \\
 & & & k_{10\,4} & & & & & & k_{10\,10} & & \\
 & & k_{11\,3} & & k_{11\,5} & & & & k_{11\,9} & & k_{11\,11} & \\
 & k_{12\,2} & & & & k_{12\,6} & & k_{12\,8} & & & & k_{12\,12}
\end{array} \right] \tag{15}$$

where,

$$k_{11} = \frac{EA_{eq-x}}{L_x}, \quad k_{22} = \frac{18EI^{(2)}_{eq-z}}{L_x^3}, \quad k_{33} = \frac{18EI^{(2)}_{eq-y}}{L_x^3}, \quad k_{44} = \frac{GC_{eq-x}}{L_x}, \quad k_{53} = \frac{E}{L_x^2}\left(3I^{(1)}_{eq-y} - 9I^{(2)}_{eq-y}\right),$$

$$k_{55} = \frac{E}{L_x}\left(\frac{1}{2}I^{(0)}_{eq-y} - 3I^{(1)}_{eq-y} + \frac{9}{2}I^{(2)}_{eq-y}\right), \quad k_{62} = \frac{E}{L_x^2}\left(-3I^{(1)}_{eq-z} + 9I^{(2)}_{eq-z}\right),$$

$$k_{66}=\frac{E}{L_x}\left(\frac{1}{2}I^{(0)}_{eq-z}-3I^{(1)}_{eq-z}+\frac{9}{2}I^{(2)}_{eq-z}\right),\quad k_{71}=-\frac{EA_{eq-x}}{L_x},\quad k_{77}=\frac{EA_{eq-x}}{L_x},\quad k_{82}=-\frac{18EI^{(2)}_{eq-z}}{L_x^3},$$

$$k_{86}=\frac{E}{L_x^2}\left(3I^{(1)}_{eq-z}-9I^{(2)}_{eq-z}\right),\quad k_{88}=\frac{18EI^{(2)}_{eq-z}}{L_x^3},\quad k_{93}=\frac{18EI^{(2)}_{eq-y}}{L_x^3},\quad k_{95}=\frac{E}{L_x^2}\left(-3I^{(1)}_{eq-y}+9I^{(2)}_{eq-y}\right),$$

$$k_{99}=\frac{18EI^{(2)}_{eq-y}}{L_x^3},\quad k_{104}=-\frac{GC_{eq-x}}{L_x},\quad k_{10\,10}=\frac{GC_{eq-x}}{L_x},$$

$$k_{113}=\frac{E}{L_x^2}\left(-3I^{(1)}_{eq-y}-9I^{(2)}_{eq-y}\right),\quad k_{115}=\frac{E}{L_x}\left(-\frac{1}{2}I^{(0)}_{eq-y}+\frac{9}{2}I^{(2)}_{eq-y}\right),\quad k_{119}=\frac{E}{L_x^2}\left(-3I^{(1)}_{eq-y}+9I^{(2)}_{eq-y}\right),$$

$$k_{11\,11}=\frac{E}{L_x}\left(\frac{1}{2}I^{(0)}_{eq-y}+3I^{(1)}_{eq-y}+\frac{9}{2}I^{(2)}_{eq-y}\right),\quad k_{122}=\frac{E}{L_x^2}\left(3I^{(1)}_{eq-z}+9I^{(2)}_{eq-z}\right),\quad k_{128}=\frac{E}{L_x^2}\left(-3I^{(1)}_{eq-z}-9I^{(2)}_{eq-z}\right),$$

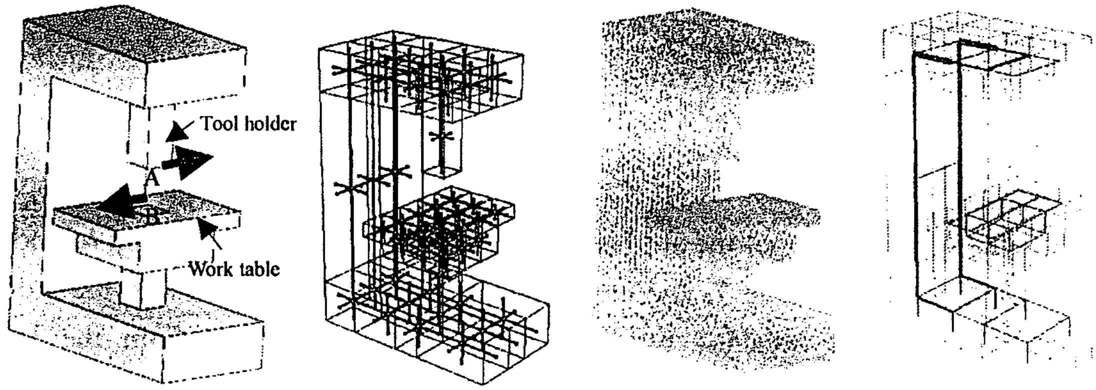
$$k_{126}=\frac{E}{L_x}\left(-\frac{1}{2}I^{(0)}_{eq-z}+\frac{9}{2}I^{(2)}_{eq-z}\right),\quad k_{12\,12}=\frac{E}{L_x}\left(\frac{1}{2}I^{(0)}_{eq-z}-3I^{(1)}_{eq-z}+\frac{9}{2}I^{(2)}_{eq-z}\right)$$

By the same way, we can get the stiffness matrices for the other combination of nodes, such as the node 2 and node 7, and etc. Then, the total stiffness matrix in the global coordinates system is obtained by the coordinate transformation from the local coordinates by the usual manner.

3. Numerical Analysis by Proposed Structural Element Method

The proposed 3D-FOA method described in the previous section has been applied to analyze several typical shaped examples. We have performed the simulation of deformation for different typical model. Among them one of the complicated example is described in the following. The Young's modulus E is assumed as 200 GPa through the examples. The results obtained by the proposed method are compared with the results estimated by the elementary theory as well as the results obtained by the FEM. Then, the validity of proposed FOA method is discussed.

We have applied the proposed 3D-body FOA method to a kind of real machine tool structures like a milling machine as shown in Fig.5 (a). The rough sizes of the model are 2.0 m in height, 1.0 m in width and 1.5 m in depth. The model is fixed on the bottom surface and is applied a pair of cutting forces 4.0×10^3 N in the opposite directions at the center point A of tool holder and a center point B of work table as shown in the figure. The milling machine model structure has been descritized into 43 3D-body FOA elements as shown in Fig.5 (b), and the deformation at the loading points are compared with the results of usual FEM using the mesh of Fig.5(c). The displacements in the force direction at A and B are obtained as 1.75 mm and –1.17 mm by the FOA, and 1.09 mm and –1.62 mm by the FEM, respectively. The percentages of error at A and B are 60.5% and 27.8%. Fig.5(d) shows the axial element forces distribution along the FOA elements, in which the values of axial forces are indicated by the thickness of solid lines. From this figure, it is seen that the bending moment caused by the cutting force at the tool holder is transmitted by a pair of tensile and compressive axial forces along the backside column to the base structure. It is noticed that the proposed 3D-body FOA method can estimate the rough deformation and the loading path with small number of FOA elements.



(a) Loading condition (b) FOA model (c) FEM model (d) Axial loading paths

Fig. 5 Structural model of milling machine

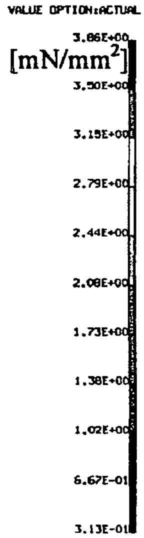
Even though the FOA formulation of 3D-body has been established, some issues such that how to divide the models into the sub-regions of 3D-blocks, how to place the nodes of FOA elements and how to construct the proposed beam-type FOA elements as well as the precision of the FOA method are still remained. I expect to discuss as the future works.

4. Formulation of Stress Concentration Prediction

Now a day it is widely recognized that the stress concentration affects the lifetime of machine products under tension, bending, and/or shear, and that the stress concentration factor is of paramount importance for assessing fatigue strength in notched parts.

A cylinder with deep U-notch shown in Fig.6 is considered as an example. Because of symmetry of the model, 1/8 and a quarter models for tensile and bending loads and full model for shear and twisting load are considered, respectively, and approximated formulas are rived as functions of design variables. By keeping the diameter D of the cylinder constant, the ratios of U-notch depth to the diameter $x_1= 2d/D$ and root radius to the diameter $x_2 = 2\rho /D$ are taken as the design parameters. The tensile and shear forces, and bending moment as well as twisting moment are applied at the ends and the upper and lower bounds (changeable ranges) are assigned for each design parameters as

$$0.20 \le x_1 \le 0.40, \qquad 0.030 \le x_2 \le 0.150,$$



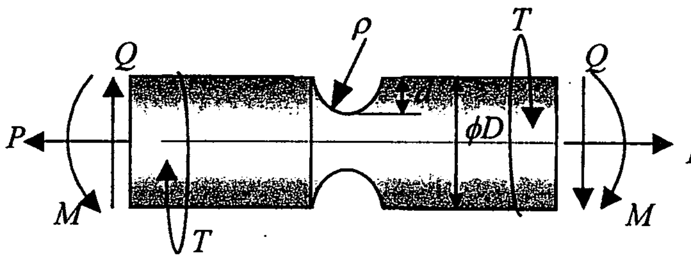
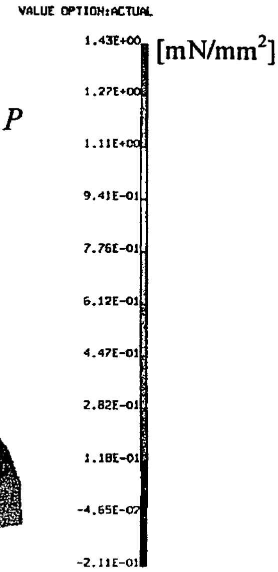


Fig.6 A solid bar with U-notch



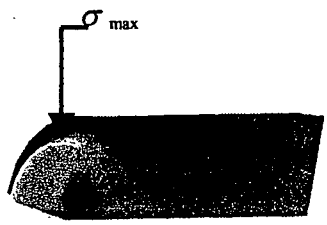


Fig.7 Deformation under tensile force.

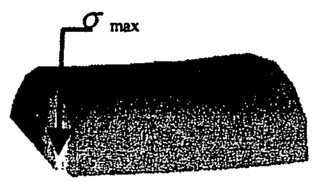


Fig.8 Deformation under bending moment.

The approximate formula of maximum stress is constructed by the polynomial regression. 9 sampling points of design parameter combination are analyzed, and the stress concentration factor for tensile, bending, shear and twisting load are calculated by keeping all other parameters constant. Figures 7 and 8 show the deformations and stress distributions for tension and bending moment. It is found that the maximum stresses occur at the bottom of surrounding notch portion for tensile and twisting load, however for shear and bending load it occurs a point. Finally, the approximate formulas are obtained as follows:

(1) Tensile force:

$$\sigma_{\max}/\sigma_{ao} = 3.4819 + 12.4958x_1 - 11.8754x_2 - 16.6550x_1^2 + 10.8181x_2^2 - 39.5939x_1x_2 + \varepsilon$$

(2) Bending force:

$$\sigma_{\max}/\sigma_{b0} = 2.9006 + 6.9710x_1 - 8.4533x_2 - 12.0235x_1^2 - 16.9534x_2^2 - 9.8178x_1x_2 + \varepsilon$$

(3) Shear force: (16)

$$\tau_{\max}/\tau_{co} = 3.4689 + 2.1287x_1 - 11.3619x_2 - 8.1378x_1^2 - 5.7140x_2^2 + 9.2948x_1x_2 + \varepsilon$$

(4) Twisting force:

$$\tau_{\max}/\tau_{d0} = 1.9389 + 3.4569x_1 - 1.1190x_2 - 4.8962x_1^2 - 3.9723x_2^2 - 17.8172x_1x_2 + \varepsilon$$

where nominal stresses are $\sigma_{a0} = P/\pi r^2$ for tensile force P, $\sigma_{b0} = 4M/\pi r^3$ for bending moment M, τ_{c0} =, $Q/\pi r^2$ for shear force$Q$, and $\tau_{d0} = 2T/\pi r^3$ for twisting moment T, in which r denotes radius of the cylinder at notch portion of minimal cross section.

To prove the reliability and effectiveness, the derived formulas have been applied for the case $2d/D = 0.225$, and $2\rho/D = 0.1$, when the nominal stress ratios of tension to bending moment $\sigma_{a0}/\sigma_{b0} = 1.9375$, and of tension to twisting moment $\sigma_{a0}/\tau_{d0} = 3.875$ are applied. Then, the maximum stress at the bottom of notch portion is predicted from our formula as $\sigma_{\max}/\sigma_{a0} = 5.3291$, whereas the direct CAE analysis gives $\sigma_{\max}/\sigma_{a0} = 4.65$, and the error is 14.60%. So it is found that the prediction by suggested formulas is reliable and useful for the FOA of complicate continuum structures in 3-dimension.

5.Conclusions.

In this thesis paper, a beam type 3D-body First Order Analysis element as an assembly of six beam elements is proposed to analyze the deformation and the loading path of complicate three-dimensional structures, and the equivalent stiffness of axial, bending and torsional deformation have been formulated. Then the proposed FOA element has been applied to analyze the deformation of several kinds of typical structures. And the approximate formulas of stress concentration factor of element with notch for three-dimensional model has been constructed, and validity of the formulas have been confirmed numerically by applying to arbitary ratio of combinatorial loading cases. From these numerical results, it is found that the proposed 3D-body FOA method can estimate the approximated global deformation and stress concentration factor of three-dimensional complicate structures roughly, that are useful for considering the load transmitting path of the structure and for designing a better structure by changing the stiffness of structural components.

References

(1) Lemon, J.R., Tolani, S.K. and Klosterman, A.L., "Integration and Implementation of Computer Aided Engineering and Related Manufcaturing Capabilities into Mechanical Product Development Process", Gi-Jahrestagung, (1980).

(2) Nishigaki, H., Nishiwaki, S., Amago T. and Kikuchi, N., "First Order Analysis for Automotive Body Structure Design", *Proc. of ASME DETC*, DAC 14533(2000), Maryland.

(3) Nishigaki, H., "First Order Analysis for Automotive Body Structure Design Using Excel", *R&D Review of Toyota CRDL*, Vol.37, No.1(2001), pp.1-8.

(4) Balabanov, V. and Venter, G., "Multi-Fidelity Optimization with High-Fidelity Analysis and Low-Fidelity Gradients", *10th AIAA/ISSMO Multidisciplinary Analysis and Optimization Conference*, AIAA Paper 2004-4459(2004), Albany, NW.

(5) Eldred, M. and Dunlavy, D., "Formulations for Surrogate-Based Optimization with Data Fit, Multifidelity, and Reduced-Order Models", *11th AIAA/ISSMO Multidisciplinary Analysis and Optimization Conference*, AIAA Paper 2006-7117 (2006), Portsmouth, VA.

(6) Microsoft Corporation, *Microsoft Excel/Visual Basic Reference*, 2nd Ed. (1998), Microsoft press.

(7) Bathe, K. J., *Finite Element Procedures*, (1995), Prentice-Hall, Englewood Cliffs.

(8) Washizu, K., et al. ed., *Handbook of finite element method*, (1981),

学位論文審査結果の要旨

当該学位論文に関し，平成 20 年 1 月 29 日に第 1 回学位論文審査委員会を開催し，提出された学位論文および関連資料について検討を加え，2 月 6 日の口頭発表後，第 2 回審査委員会を開催し，協議の結果以下の通り判定した．

本論文は，構造設計上流部の概念設計段階で有用な設計ツールとなりうる三次元 FOA(First Order Analysis)の具体的な簡易解析法として，三次元はり要素を組合わせた FOA 近似解析要素を提案し，その具体的な引張・圧縮，曲げ，ねじりの等価剛性導出法を定式化したものである．また基本的な力学モデルや三次元組立てモデル，簡易工作機械モデルに適用してその有効性を示した．さらに FOA 解析結果から得られる軸力や曲げモーメント，ねじりモーメント，せん断力から，三次元部品内に内在する切欠きや穴，段差に起因する応力集中を応答曲面近似によって予測する具体的な方法を提案し，各種応力集中モデルに適用して近似度についても検討を加えている．これらの基本的な検討結果は，将来的な FOA 設計ツールの開発に対する明るい見通しを与える先駆的な業績と位置付けられる．

以上より本論文は，将来の三次元物体の FOA 解析・設計ツール開発に対する道を切り拓くもので，博士（工学）の学位に値するものと判定した．